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A method for the recovery of waste heat energy exhausted by a main propulsion diesel engine and ships service electrical power diesel engines in a naval vessel is proposed and evaluated. Waste heat recovery boilers and steam system components are designed for the recovery of the exhausted diesel energy as a means of improving the fuel efficiency of diesel engines for shipboard application. The waste heat recovery analysis evaluates the main propulsion diesel engine operating at 50% maximum continuous rating and ships service electrical power diesel engines operating at 75% maximum continuous rating. Waste heat boilers and steam system components are sized and a possible propulsion plant arrangement is presented.

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DIESEL EXHAUST WASTE HEAT RECOVERY FOR NAVAL VESSELS

by

JAMES RUSSELL DIXON

B.S. Mech. Eng., Purdue University
(1974)

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DIESEL EXHAUST WASTE HEAT RECOVERY
FOR NAVAL VESSELS

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JAMES RUSSELL DIXON

Submitted to the Department of Ocean Engineering on May 8, 1981, in partial fulfillment of the requirements for the Degrees of Ocean Engineer and Master of Science in Mechanical Engineering.

ABSTRACT

A method for the recovery of waste heat energy exhausted by a main propulsion diesel engine and ships service electrical power diesel engines in a naval vessel is proposed and evaluated. Waste heat recovery boilers and steam system components are designed for the recovery of the exhausted diesel energy as a means of improving the fuel efficiency of diesel engines for shipboard application. The waste heat recovery analysis evaluates the main propulsion diesel engine operating at 50% maximum continuous rating and ships service electrical power diesel engines operating at 75% maximum continuous rating. Waste heat boilers and steam system components are sized and a possible propulsion plant arrangement is presented.

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CHAPTER 1

INTRODUCTION [1]

Over the last decade the spiralling cost increases of marine fuel oils has had a dramatic effect on the marine industry. The cost of marine propulsion fuel has dictated marine transportation costs. For example, in 1967 bunker oil costs represented 28% of total merchant ship operating expenses while in 1976 the figure had risen to 45%. This is largely because diesel fuel costs have climbed from $27 per ton in 1969 to over $150 per ton in 1976. Present bunker oil costs have further increased to $350 per ton.

This environment has encouraged the diesel engines selection as the prominent marine main propulsion source. In 1978 diesel engines were installed in 96.8% of all newly constructed merchant ships which exceeded 2000 tons displacement. This clearly reflects the superiority of the diesel engine as a fuel efficient main propulsion.

It is, however, surprising that a large fraction of the fuels energy is not converted into mechanical energy for main or auxiliary propulsion. In fact, only about 37% of the fuels energy is extracted by conventional means for use by a turbocharged engine operating at full load. The remainder of the energy was previously exhausted, hence lost. However, as the cost of energy has increased it has become commercially

feasible to recover and use a greater portion of this exhausted energy.

The greatest portion of the exhausted energy from a diesel engine has been the exhausted combustion gas. These gases carry with them 36% of the fuels energy from a turbo-charged diesel engine operating at full load. Obviously the recovery of all or some portion of the exhausted energy from the diesel engine will improve the fuel efficiency of the diesel and consequently its commercial performance.

The impact of higher propulsion fuel costs has been realized by the military as well as the private commercial sectors. It is recognized that the cost of operating a combatant is 50% of the life cycle cost of a naval combatant vessel. As the cost of fuel rises more rapidly than other life cycle cost factors the percentage of life cycle costs attributable to fuel will increase even further.

The advantages which the diesel engine provides for service aboard a naval combatant encourage its acceptance as a main propulsor and prime mover for electrical power generation. Diesels are characterized by relatively low fuel consumption throughout the entire operating range. This feature is advantageous due to the varied operating speed profile of the naval surface combatant. The diesels fast response and start up time is a disirable feature for naval service application.

More modern combatant designs have become volume limited due to greater requirements for low density electronic equipments installed. The need for more space within the combatant encourages the selection of a more fuel efficient design requiring less fuel storage volume. A method of recovery of exhausted energy would further encourage volume reduction for fuel stowage.

A propulsion scheme for naval combatant application using diesels, incorporating waste heat recovery, provides the naval architect with a design which is compatable with the volume demands of modern electronic equipment installation. This study will evaluate a design for diesel engine exhaust waste heat recovery for a naval surface combatant. The propulsion and auxiliary machinery will include a medium speed diesel main propulsor and three medium speed diesel driven ships service electrical power generators.

The study will evaluate waste heat recovery and develop a design rationale which will best integrate the waste heat equipment within ship system architecture.

CHAPTER 2

DIESEL PRIME MOVERS

2.1 Main Propulsion Diesel Engine [13]

A medium speed diesel engine is selected as the main propulsion engine for the surface combatant. The medium speed diesel is chosen for its compact size when compared with slow speed diesels of comparable output power. The specific engine selected is the Colt-Pielstick, 18 cylinder, type PC4 diesel engine. This engine is capable of producing 27,000 horsepower which is suitable for the propulsion demands of a surface combatant.

The engine weighs 369 tons, has an overall length of 43 feet 10 inches, a height of 21 feet 8 inches, and a width of 14 feet 8 inches.

For the purpose of evaluating a waste heat recovery system for the diesel engine the assumed operating power level for the diesel engine will be 50% of its maximum continuous rating. This value is chosen because surface combatants rarely operate at maximum speed.

2.2 Ships Service Diesel Generator [14]

The diesel prime mover selected for the generation of ships electrical power is a medium speed diesel engine. The electrical power requirements of modern combatants have increased substantially due to the installation of more

electronic equipments. Current designs require as much as 6000 Kw of electrical power in the most demanding situations.

The diesel prime mover chosen for ships service power generation is the Pielstick PA6V type 280, 12 cylinder diesel. This engine is capable of producing 3271 Kw of electricity, assuming a 95% generator efficiency. The engine operates at 900 rpm when producing 60 Hz electrical power.

It is assumed that two diesel generator sets are sufficient to provide power for the most demanding situation aboard ship. A third generator set will be included to provide sufficient redundancy for continued electrical power generation.

For the purpose of evaluating a waste heat recovery system for these diesel engines it is assumed that a single diesel operating at 75% of the maximum continuous rating is sufficient to supply normal underway electrical requirements. This will allow a sufficient margin for transient power surges and is a feasible amount of power to provide normal underway power capability. For maximum electrical demands as would occur when all electronic equipments are operating it is assumed that two diesel generating sets, each operating at 75% of the maximum continuous rating, are sufficient to supply all electrical power demands.

CHAPTER 3

AVAILABILITY AND UTILITY OF WASTE HEAT ENERGY

3.1 Available Waste Heat Energy

The diesel engine converts only some of the energy of the combusting fuel into power. A larger portion of the fuels energy is rejected by the engine. In an attempt to recover some of this rejected energy it becomes necessary to evaluate and quantify the energy in the diesel engine.

Due to the above ambient temperature of the diesel cycle there is some energy lost due to radiation from the engine to the surroundings which are at a lower temperature. There are other mechanisms of heat transfer; namely convection and to some lesser extent conduction which account for heat transfer and subsequent energy loss. Though all three mechanisms of heat transfer occur the loss of energy due to the relatively higher temperature of the engine is referred to as radiation loss.

Diesel engines are turbocharged engines vice air introduction through natural aspiration. The motive power required for turbocharging is extracted from the combustion gases exhausted from the engine. These gases are expanded through a power turbine which is coupled to a compressor which delivers a fresh air charge to the combustion chamber at above ambient pressure. The turbocharging process does not

extract energy from the engine but rather from the exhausted combustion gases.

The lubricating oil in the diesel engine both lubricates and convects heat from the diesel engine internal surfaces. The heat energy is subsequently removed from the oil as it passes through a cooler. This necessary convection of heat from the internals of the diesel engine represents another source of exhausted energy.

Energy is also exchanged between engine jacket cooling water and the diesel engine. The jacket cooling water convects heat away from the cylinder walls so as to preclude adverse thermal effects on the heat affected zones in the engine. This water is subsequently cooled for reuse in the engine to again convect heat from the vicinity of the cylinder walls. The jacket cooling water cycle removes energy from the diesel engine.

An air cooler is employed in the charge air system to remove heat from turbocharged air prior to injecting the air into the combustion chamber. The cooled air is more dense following cooling consequently a greater mass of air is introduced into the combustion chamber which is advantageous for engine performance. Heat is transferred from the air charge to water in the air cooler. This water is subsequently cooled for reuse whereby energy is removed and again rejected.

The greatest source of rejected energy from the diesel engine is the exhausted combustion gases. These gases are the necessary products of combustion of the burnt fuel air mixture from the combustion chamber.

A thermal balance for the ships service diesel engines is given in Table 3.1. The data is normalized as a percentage of total energy in the cycle. The conditions are for a fuel of lower heating value of 18,000 Btu/lbm.

Diesel Engine Maximum Continuous Rating	Percent of Total Energy 100%	75%	50%
Jacket Water & Air Cooler	10	11	12
Turbocharger	9	7	6
Oil	4	5	6
Exhaust Gas	34	33	34
Radiation	2	2	2
Power	41	42	40

Table 3.1

A thermal balance for the main propulsion diesel is presented in the Table 3.2. The conditions are for a fuel of lower heating value of 18,000 Btu/lbm.

Diesel Engine Maximum Continuous Rating	Percent of Total Energy 100%	75%	50%
Jacket Water & Air Cooler	7	8	8
Turbocharger	10	8	7
Oil	4	3	3
Exhaust Gas	32	32	33
Radiation	2	2	3
Power	45	47	46

Table 3.2

For both the main propulsion diesel engine and ships service electrical power diesel engine less than half of the fuels energy is extracted as useful work. The power which is recoverable is actually the indicated horsepower and is subject to transmission losses in converting it to useful work. The percentage power shown is actually the maximum attainable before transmission losses are considered.

Before an assessment can be made of the recovery of waste heat energy it becomes necessary to determine the quality of the exhausted energies.

For a single ships service diesel engine operating at 75% of its maximum continuous rating the exhausted energies are quantified in Table 3.3.

Source of Energy	Flow Rate	Temperature
Jacket Cooling Water	594 gpm	185°F
Charge Air & Lube Oil Cooling Water	145 gpm	172°F
Exhausted Combustion Gas	11.4 lbm/sec	707°F

Table 3.3

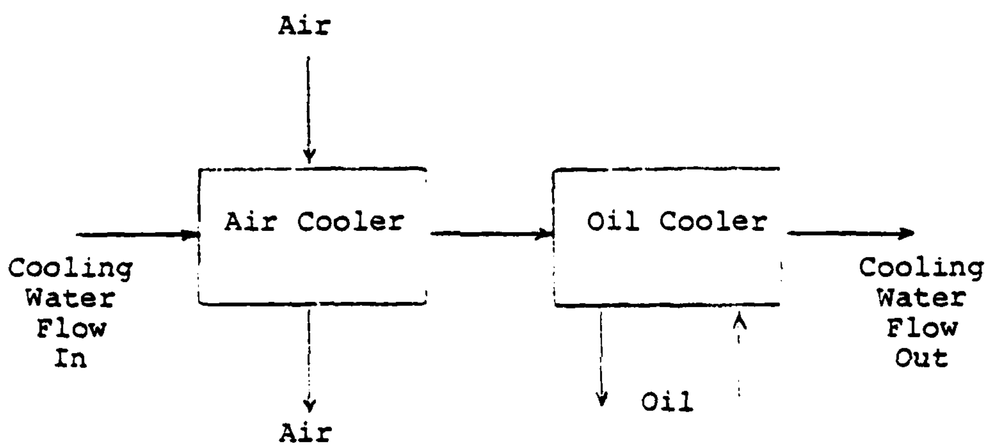
For the main propulsion diesel engine operating at 50% of its maximum continuous rating the exhausted energies are quantified in Table 3.4.

Source of Energy	Flow Rate	Temperature
Jacket Cooling Water	2512 gpm	185°F
Charge Air & Lube Oil Cooling Water	613 gpm	172°F
Exhausted Combustion Gas	46.74 lbm/sec	692.6°F

Table 3.4

Radiation heat transfer is excluded from the analysis of energy recovery because of the comparatively low amount of energy lost due to radiation. There would be no simple mechanism to recover this energy from the engine worthy of consideration. The energy associated with the charge air and

lube oil cooling water was coupled in Tables 3.3 and 3.4 due to the series arrangement for cooling water flow. This arrangement is presented in Figure 1 and is used with the diesel engines.



<u>Figure 1</u>

In evaluating the availability of energy in the various heat rejection processes the analysis is to determine the amount of reversible work that can be done in a process that exchanges heat only with the surroundings. It depends on the difference between its temperature and that of ambient air or water which constitute the surroundings. If a conservative estimate is made of ambient temperature such that ambient

temperature is 100^{o}F an analysis can be conducted as to the utility of energy recovery.

The charge air and oil cooling water outlet temperature of 172^{o}F coupled with a flow rate were below that of jacket cooling water substantially raduce the utility of energy recovery for this application. The jacket cooling water represents a greater amount of exhausted energy due to a slightly higher temperature and significantly greater flow rate than the charge air and oil cooling water. This energy is similarly less attractive for recovery due to its relatively lower temperature than the exhausted combustion gases. Recovery of some of this low quality energy, so called in literature, is most successfully accomplished when combined with exhaust gas recovery methods. The most widely practiced method is with ebullient heat recovery. This method involves heat exchange between combustion gases and the jacket cooling water. The jacket cooling water boils and provides low pressure steam for various services. This method of heat recovery has not found wide application in larger diesel installations.

Another potential method of recovery of this energy would be to use the jacket cooling water to preheat feed water in a boiler "fired" by exhausted combustion gas. This method has found application in shore based power generating stations where space requirements are not as demanding as in

shipboard application. To allow an appreciable amount of energy transfer requires substantial heat transfer surface area due to the low temperature differential between the two fluids.

The exhausted combustion gas represents the greatest source of waste heat energy and is most attractive due to its high temperature which allows greater energy transfer in a more compact volume. By withdrawing energy from the exhaust gas stream the temperature of the discharged gas from the stack will be reduced. This is advantageous for a naval vessel due to reduced infrared emissions thereby reducing detection.

3.2 Utility of Waste Heat Energy

Given that there is available waste heat energy from the main propulsion and ships service electrical power diesel engines it becomes necessary to evaluate the requirements for energy aboard the ship. Though shipboard energy requirements are not constant due to the posture of the vessel there are two conditions which specify general energy demands. There are shipboard steam requirements for cruising operating conditions which specify normal steam demands for the ship while underway. More demanding steam requirements are specified as full power or maximum load conditions. These requirements are presented in Table 3.5 and are representative of a destroyer type naval vessel.

Requirement	Cruising			Full Power		
	Pressure (psig)	Temp (^{o}F)	Flow Rate (lbm/hr)	Pressure (psig)	Temp (^{o}F)	Flow Rate (lbm/hr)
Galley, Laundry, Hot Water	150	400	1400	150	400	1400
Fuel Oil Heating	150	400	225	150	400	700
Evap Air Ejector	150	400	600	150	400	600
SSTG Air Ejector	150	400	180	150	400	180
Ships Heating	---	---	---	150	400	1400
Evaporators	14	272	50	16	304	50

Table 3.5

It is desired that the waste heat energy recovered be applied in a form compatable with existing shipboard energy requirements. The maximum steam pressure (150 psig) and temperature (400°F) dictated by shipboard requirements will serve as the minimum design parameters in application of waste heat recovery. This will insure minimum impact on existing ship design energy requirements.

Because shipboard steam requirements vary it becomes necessary to channel waste heat energy toward other applications. This would allow consumption of all waste heat energy and consequent improved fuel efficiency of the diesel engine and waste heat recovery system. Another energy requirement aboard ship is electrical power. If waste heat energy could be applied toward power generation the requirement for diesel supplied electrical power could be reduced which implies a more energy efficient configuration.

Because steam is supplied for shipboard services, excess steam could be used to generate electrical power by use of a steam turbine driving a generator. The condenser for this steam turbine could be used to condense both exhausted steam from the turbine and steam returns from shipboard steam services.

CHAPTER 4

EXHAUST GAS WASTE HEAT RECOVERY

4.1 Exhaust Gas Side of Waste Heat Boiler

4.1.1 Waste Heat Boiler Gas Side Fouling [2]

Diesel exhaust gas waste heat boilers have characteristically been of a finned tube design on the gas side of the boiler. The finned tube allows greater heat transfer, hence greater energy recovery per unit of tube length. For fin spacing of 5 fins per inch significant fouling of heat transfer surfaces occurs. This fouling is evident as the annulus between fins becomes plugged with particulate derived from the combusted diesel fuel ash. When fin spacing is increased such that there are 3 fins per inch no plugging occurs.

Finned tube fouling in the diesel exhaust gas side of the boiler is temperature dependent. Tubes operating at a mean temperature of approximately 550°F do not foul significantly whereas tubes operating between 150°F and 300°F undergo significant fouling. Evidence suggests that phenomena other than acid condensation are responsible for the majority of fouling (perhaps condensation of an unknown hydrocarbon contained in the exhaust).

4.1.2 Waste Heat Boiler Gas Side Corrosion [3,4,5,6]

In order to determine the mechanism of corrosion on the heat transfer surfaces of waste heat recovery boilers, there

has been extensive analysis of the composition of the diesel engine exhaust gases. It has been determined that the presence of sulfuric anhydride and water vapors in the combustion products causes the formation of sulfuric acid vapors that condense on the surfaces having lower temperatures. Under the action of sulfuric acid in solution the low temperature heating surfaces will deteriorate. Therefore, the main source of heat exchanger corrosion in diesel exhausts is from the formation of sulfuric acid from the oxidation of initial fuel sulfur.

The U.S. Navy standard for marine diesel fuel prohibits sulfur concentration from exceeding 1%. This may be considered a maximum practical sulfur concentration as sulfur content in commercially available fuel varies between 0.05% and 0.80%.

The potential for sulfuric acid condensing on heat transfer surfaces and corroding the material can be inhibited by maintaining heat transfer surface temperature above the condensing temperature of the sulfuric acid vapor. The acid dew point temperature is approximately 285°F. It has become a design practice to insure that the lowest exhaust gas temperature does not fall below 300°F. In the design of a waste heat boiler this limit will be employed to reduce the potential for acid corrosion of the exhaust gas side heat transfer surfaces.

Selection of a tube and fin material which is more resistant to acid corrosion also insures reduced corrosion potential. Besides corrosion durability, the selection is

tempered by consideration of heat transfer performance, temperature gradients, thermal shock, strength as a pressure boundary, cost and ease of fabrication. Common materials for this application include carbon steel (SAE 1020), stainless steel (316) and copper bearing steel (Corten). These materials give the most acceptable performance when the previous criteria are considered. Carbon steel is the least expensive material to use for finned tube construction and will be used for the waste heat boiler tubes.

4.1.3 Diesel Exhaust Gas Pressure Drop Across the Waste Heat Boiler [7,2]

Diesel engine performance can be adversely affected by excessive exhaust gas back pressure. Specifications for the Pielstick diesel engines used for main propulsion and electrical power generation prescribes a maximum of 10 inches of water pressure as an acceptable pressure differential through exhaust ducts. In sizing the waste heat boiler a maximum pressure drop across the boiler of 6 inches of water pressure differential will be allowed. This will permit a maximum pressure differential across ducting of 4 inches of water pressure.

A cross-counter flow heat exchanger arrangement will be employed, as shown in Figure 2.

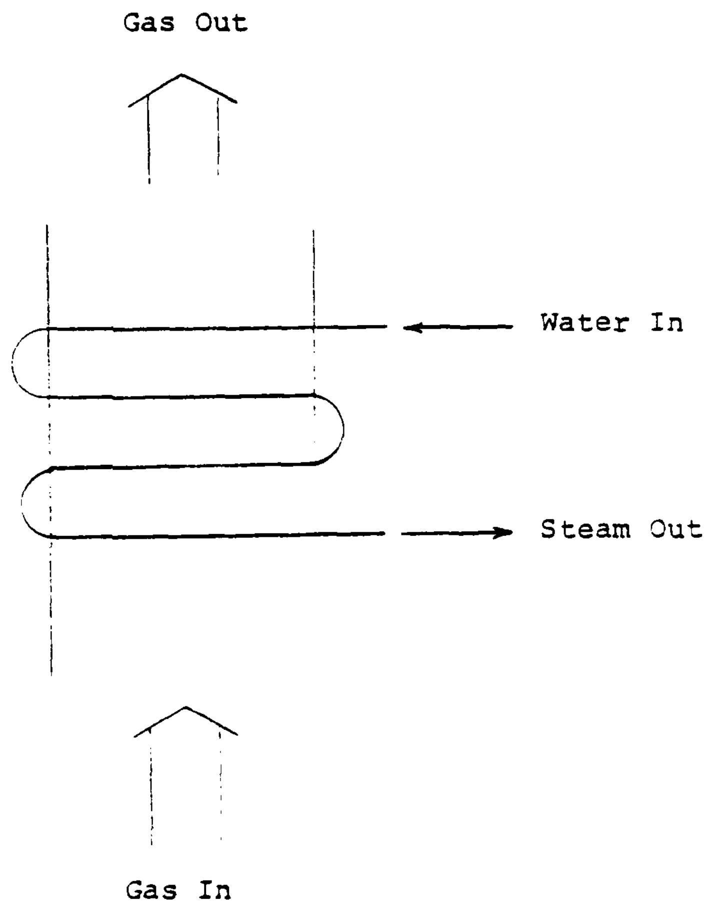


Figure 2

The tube arrangement in the boiler will be as repre-sented in Figure 3.

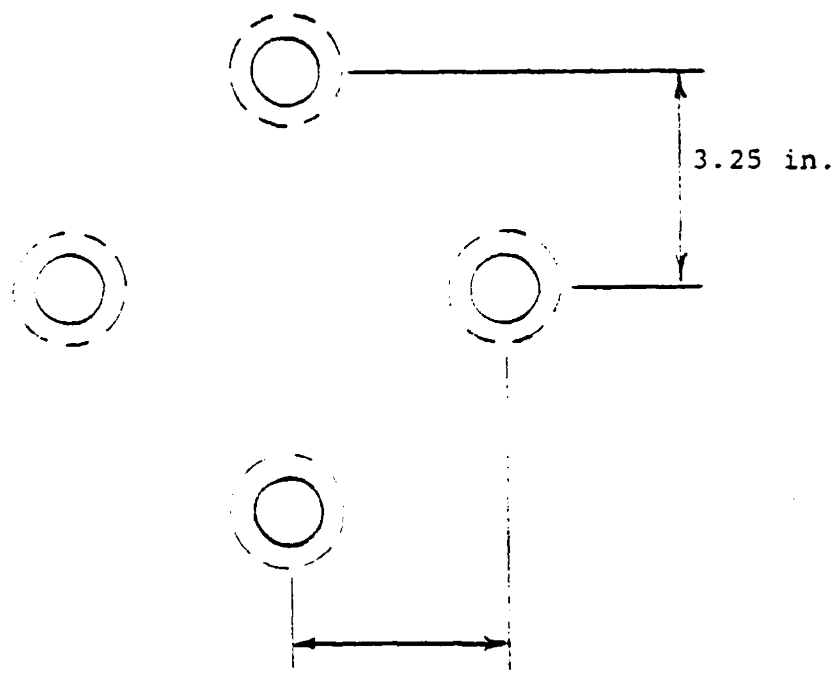


Figure 3

The boiler tubes specifications are as follows:

Tube and Fin Materials	1020 carbon steel
Tube Outside Diameter	1.64 inches
Fin Height	0.75 inches
Fin Thickness	0.06 inches
Number of Fins Per Inch	3.0

Pressure drop characteristics of a heat exchanger employing the tube arrangement shown in Figure 3 and using the tube and fin characteristics previously specified are given in Figure 4.

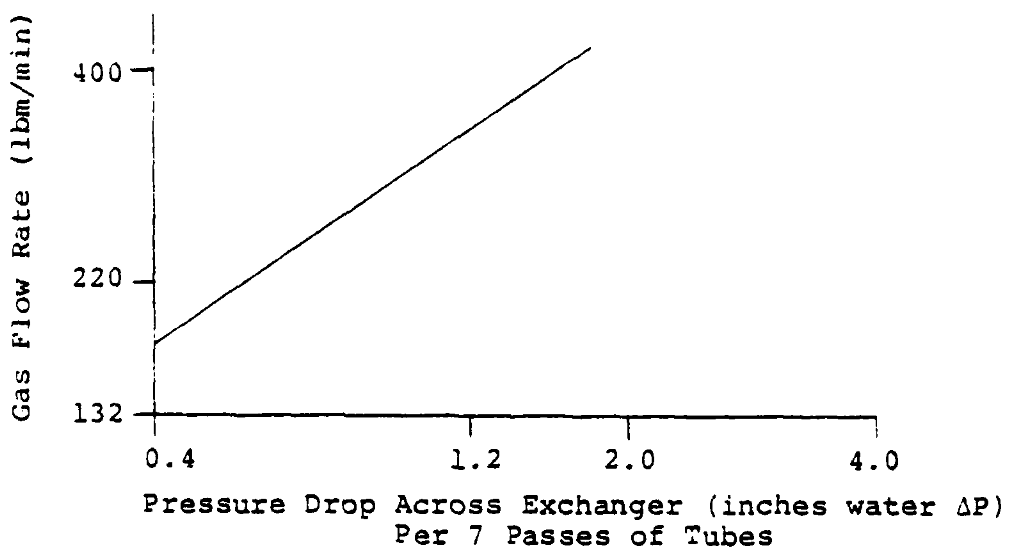


Figure 4

The heat exchanger employed in deriving the characteristics of Figure 4 employed a 7 pass cross-counter flow arrangement and had an area of 4.15 square feet normal to the diesel exhaust gas flow.

The following procedure will allow sizing of the waste heat boiler for diesel exhaust gas pressure differential.

$P \equiv$ number of passes of steam and water flow through the boiler

$N = P/7$

N = number of passes per seven tube increment

$P' = 6$ inches water pressure/N

P' = allowable pressure drop (inches water pressure) per seven tube group, this value is the abscissa of Figure 4.

Using Figure 4 the allowable air flow rate per 4.15 square feet (area normal to exhaust gas path) is determined.

$A = 4.15\ \dot{m}$

$\dot{m}$ = gas flow rate (lbm/min), from Figure 4

A = area normal to exhaust gas path.

4.2 Steam Side of the Waste Heat Boiler

4.2.1 Sizing the Boiler for Heat Transfer [8,9]

The amount of energy extracted from the diesel engine exhaust gas stream will determine the area required for heat transfer and ultimately determine the waste heat boiler dimensions. In order to determine the heat transferred from the exhaust gases to the boiler water and steam it will be necessary to evaluate the flow regimes in the boiler.

Figure 5 illustrates the temperature relation between diesel exhaust gas and the boiler water and steam mixture. Within the boiler, water can be generally classified into four flow regimes. In area A (Figure 5) the boiler water is a compressed liquid and is heated to the saturated liquid state at point 3. From point 3 to 4 (area B) the water starts as a saturated liquid and becomes, with heat addition, a 70% quality water-steam mixture at point 4. This is called the low boiler region where boiling heat transfer predominates. In region C the 70% quality steam is heated to the saturated vapor state at point 5. The flow regime is called mist flow where most of the heat transfer is from the hot wall to the vapor, and after the heat has been transferred into the vapor core it is transferred to the liquid droplets there. The saturated steam becomes superheated in area D where heat is transferred from one gas (diesel exhaust) to another (superheated steam). These different flow regimes will have a

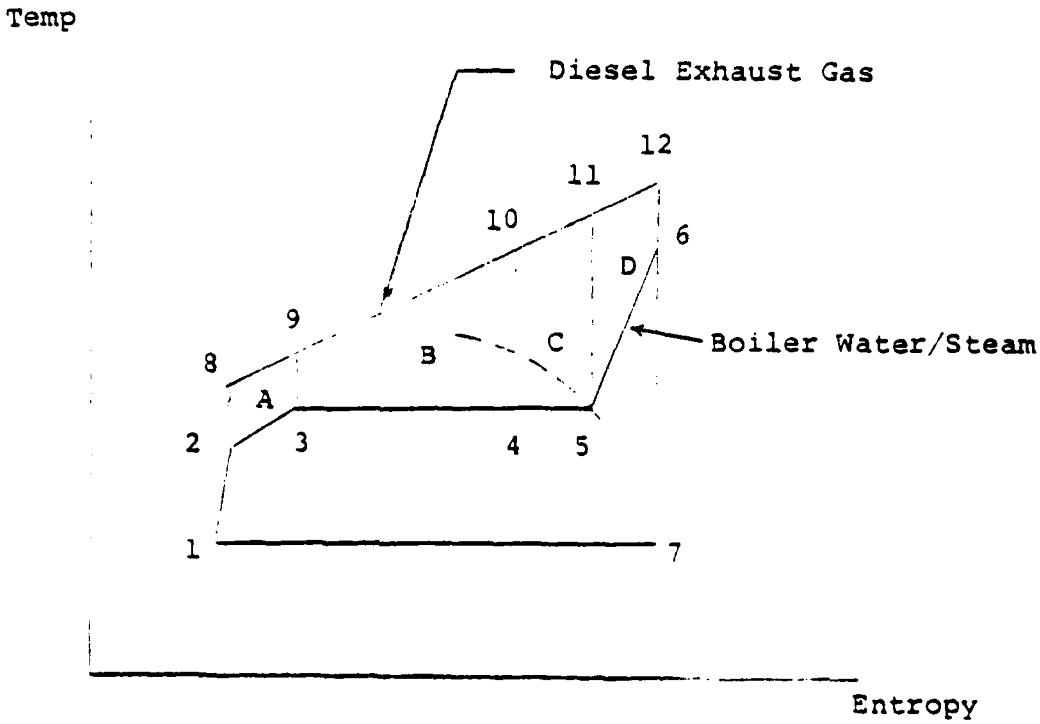


Figure 5

substantial effect on the heat transferred in each area and consequently the surface area required for heat transfer. This ultimately impacts boiler size.

For both the main propulsion diesel and the ships service electrical power diesel waste heat boilers the method of sizing the boilers will be similar.

4.2.1.1 Determining Heat Transferred to the Waste Heat Boiler

The analysis for determining heat transferred to the boiler was conducted by using determined or assumed values of T_{12}, $\dot{m}_g$ (mass flow rate of diesel exhaust gas). T_6, P_6, $\dot{m}_w$ (mass flow rate through the boiler), $\eta_{p(is)}$ (pump isentropic efficiency), $\eta_{t(is)}$ (turbine isentropic efficiency), and C_{pG} (specific heat of diesel exhaust gas).

The following are defined:

$q_{A,B,C,D}$	heat transferred in areas A,B,C, or D
T_1	temperature at point 1

The procedure is as follows:

(1) $q_D = \dot{m}_w (h_6 - h_5)$

(2) $T_{11} = T_{12} - \dot{q}_D/\dot{m}_G C_{pG}$

(3) $q_C = \dot{m}_w (h_5 - h_4)$

(4) $T_{10} = T_{11} - q_C/\dot{m}_G C_{pG}$

(5) $q_B = \dot{m}_w (h_4 - h_3)$

(6) $T_9 = T_{10} - q_B/\dot{m}_G \, C_{pG}$

(7) $\Delta h = (P_2 - P_1) v_1$

(8) $h_{2(is)} = h_1 + \Delta h$

(9) $h_2 = h_1 + \frac{1}{\eta_{p(is)}} (h_{2(is)} - h_1)$

(10) $q_A = \dot{m}_w \, (h_3 - h_2)$

(11) $T_B = T_9 - q_A/\dot{m}_G \, C_{pG}$

Check: $Q = q_A + q_B + q_C + q_D$

$T_{pp} = T_9 - T_3$

$T_{pp} \equiv$ pinch point

To achieve a more acceptable pinch point or a high Q adjust $\dot{m}_w$.

To determine the minimum thickness of boiler pressure piping the following formula may be applied (Reference 10).

$$t_m = \frac{PD}{2(SE + P_y)} + A$$

t_m = minimum allowable pipe-wall thickness, [in.]

P = maximum internal service pressure, [lb/in.^{2}]

D = outside diameter of pipe [in.]

SE = maximum allowable stress in material due to internal pressure [lb/in.^{2}]

y = a coefficient obtained from Reference 10

A = allowable for mechanical strength and corrosion from Reference 10 [in.]

To determine the area per fin for spiral fins the following approximation may be made.

$$A_f = 2\pi \left[\frac{D_o^2}{4} - \frac{D_i^2}{4} \right] + \pi (D_o)(t_f)$$

A_f = area per fin [in.^{2}]

D_o = diameter to outer fin edge [in.]

D_i = diameter of pipe to fin joint [in.]

t_f = fin thickness [in.]

To determine the outer surface area of the pipe per inch with spiral fins the following approximation is made.

$A_e = [2\pi(D)][1 - n(t_f)]$

A_e = pipe outer surface area per inch [in^2/in.]

D = pipe outside diameter, [in.]

n = number of fins per inch

t_f = fin thickness [in.]

The total outside heat transfer area per inch is then computed.

$A_t = A_e + nA_f$

A_t = total pipe and fin external surface area per inch [in^2/in.]

A_e = pipe outer surface area per inch [in^2/in.]

n = number of fins per inch

A_f = area per fin [in^2]

The pipe inside surface area per inch is computed using the following relation.

$$A_i = \pi D_i$$

A_i = pipe inside surface area per inch [in^2/in]

D_i = pipe inside diameter [in.]

The inside and outside fouling factors of the heat transfer surfaces are determined using Reference 11. The tube inside and outside surface heat transfer coefficients are determined using these fouling factors.

$$h_{fi} = \frac{1}{ff_i}$$

$$h_{fo} = \frac{1}{ff_o}$$

h_{fi} = tube inside surface heat transfer coefficient [Btu/hr ft^2 °F]

ff_i = tube inside fouling factor [hr ft^2 °F/Btu]

h_{fo} = tube outside surface heat transfer coefficient [Btu/hr ft^2 °F]

ff_o = tube outside fouling factor [hr ft^2 °F/Btu]

To determine the surface heat transfer coefficient due to diesel exhaust gas passing over the tubes the following equation is applied from Reference 12.

$$h_o = \frac{k_f}{D_o} c \left[\frac{D_o \, G_{max}}{\mu_f} \right]^{0.6} Pr_f^{1/3}$$

h_o = tube outside surface heat transfer coefficient [Btu/hr ft^2 °F]

k_f = film thermal conductivity [Btu/hr ft °F]

D_o = tube outside diameter [ft.]

c = a constant from Reference 12.

G_{max} = maximum mass flux past the outside of the tubes [lbm/hr ft^2]

μ_f = film absolute viscosity [lbm/hr ft]

Pr = film Prandtl number

This relation for determining the outside surface heat transfer coefficient is applied at each of the areas of the boiler discussed in 4.2.1. This will provide a single value for the outside surface heat transfer coefficient at each of these areas.

To determine the fin efficiency the procedure in Reference 8 is applied.

$L_c = L + t/2$

L = fin height [in.]

t = fin thickness [in.]

$A_p = (L_c)(t)$ [in.^{2}]

$r_{2c} = L_c + r_1$

r_1 = tube outside radius [in.]

Determine a value for the expression:

$$L_c^{3/2}\left[\frac{h_c}{kA_p}\right]^{1/2}$$

h_c = surface heat transfer coefficient, determined for each area of the boiler previously [Btu/hr ft^2 °F]

k = fin thermal conductivity [Btu/hr ft °F]

Using the value determined for this expression and the value r_{2c}/r_1 will allow fin efficiency to be determined (at each area for the boiler) using Figure 6.

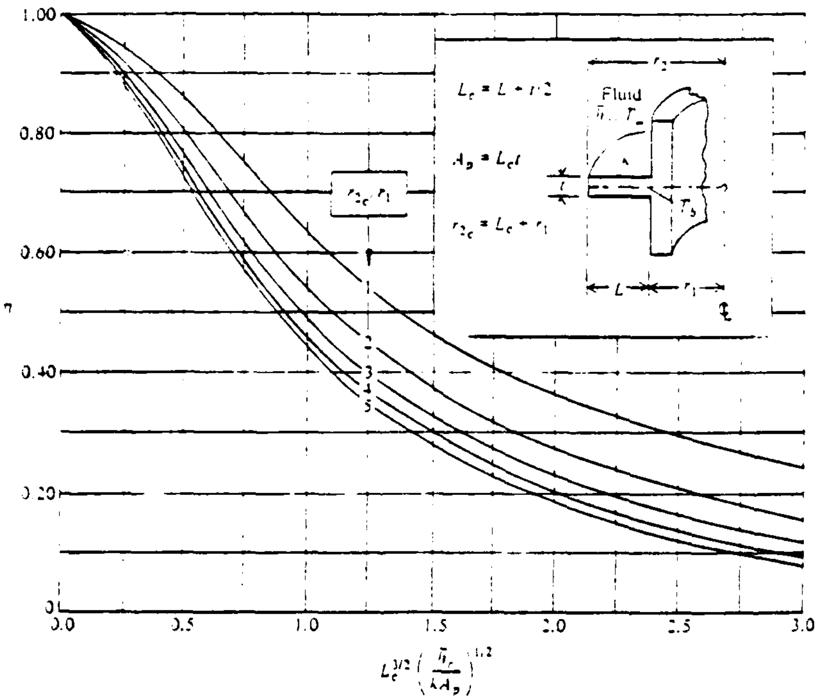


Figure 6 (from Ref. 8)

The tube and fin surface effectiveness may be determined using the following relation.

$$\varepsilon = 1 - \frac{\eta A_f}{A_t}\,(1 - \eta_f)$$

ε = surface effectiveness

A_f = area per fin [in.^{2}]

η = number of fins per inch

A_t = total outside heat transfer area per inch of tube [in.^{2}/in.]

η_f = fin efficiency

The surface effectiveness of the outside surface may be determined for each section of the boiler.

The boiler inside surface heat transfer coefficient for the compressed liquid forced convection region of area A may be determined using the McAdam's correlation in Reference 12.

$$h_i = 0.023 \frac{k_b}{D} \left[\frac{GD}{\mu_b}\right]^{0.8} \left[\frac{\mu C_p}{k}\right]_b^{0.4}$$

The conditions are evaluated at the bulk (liquid) temperature.

$$\bar{T}_b = \frac{T_2 + T_3}{2}$$

$\bar{T}_b$ = average bulk liquid temperature [°F]

T_2, T_3 as previously defined

k_b = thermal conducting of the water [Btu/hr ft °F]

D = tube inside diameter [ft]

G = mass flow rate per unit of area [lbm/ft^{2}sec]

μ_b = absolute viscosity of the water [lbm/hr ft]

C_p = specific heat of water [Btu/lbm °F]

h_2 = compressed liquid forced convection heat transfer coefficient

To determine the surface heat transfer coefficient in the low boiler (area B) it is first necessary to determine the peak heat flux in that region of the boiler. This will allow the selection of a temperature differential between tube wall and boiling liquid such that peak heat flux will not be exceeded in the low boiler. Should this peak heat flux be exceeded there is the possibility of structural damage to boiler pressure tubing.

The following relation is used to determine peak heat flux (Reference 12).

$$\frac{(q/A)_{max}}{\rho_v \, h_{fg}} = 143 \, g^{1/4} \left[\frac{\rho_\ell - \rho_v}{\rho_v}\right]^{0.6}$$

$(q/A)_{max}$ = peak heat flux [Btu/lbm ft^2]

ρ_v = density of vapor [lbm/ft^3]

h_{fg} = latent heat of vaporization [Btu/lbm]

g = acceleration due to gravity in G's

ρ_ℓ = density of liquid water [lbm/ft^3]

This relation is used to determine the peak heat flux. The value of peak heat flux is applied to the following relation to determine the temperature differential between the tube wall and boiling liquid at peak heat flux (Reference 8).

$$\Delta T_x = \frac{h_{fg} \, P_{r\ell}^{\;1.7}}{C_p} \, C_{sf} \left[\frac{q/A}{\mu_\ell \, h_{fg}} \left\{\frac{\sigma}{g(\rho_\ell - \rho_v)}\right\}^{1/2}\right]^{1/3}$$

C_p = specific heat of saturated liquid [Btu/lbm °F]

σ = surface tension of liquid vapor interface [lbf/ft]

q = acceleration due to gravity [ft/sec^2]

ΔT_x = temperature differential between tube wall and boiling liquid [°F]

Once the peak heat flux temperature differential is computed a value less than the calculated amount is selected to insure a margin of safety. To determine the surface heat transfer coefficient the following relation is applied.

$$h = \frac{q/A}{\Delta T_x}$$

h = surface heat transfer coefficient [Btu/hr ft^2 °F]

ΔT_x = temperature differential selected below peak heat flux temperature differential [°F]

q/A = heat flux corresponding to selected temperature differential [Btu/hr ft^2]

In the high quality boiler region (area C) where the mist flow governs the heat transfer in the inner tube and liquid/vapor interface the following relation is applied to determine the inner tube surface heat transfer coefficient (Reference 8).

$$h = 0.06 \frac{k_\ell}{D} \left(\frac{\rho_\ell}{\rho_v}\right)^{0.28} \left(\frac{DGX}{\mu_\ell}\right)^{0.87} P_{r\ell}^{\ 0.4}$$

h = surface heat transfer coefficient [Btu/hr ft^2 °F]

k_ℓ = liquid thermal conducting [Btu/hr ft °F]

D = pipe inner diameter [ft]

ρ_ℓ = liquid density [lbm/ft^3]

ρ_v = vapor density [lbm/ft^3]

G = mass flux through the pipe [lbm/ft^2 sec]

X = average vapor quality

μ_ℓ = liquid absolute viscosity [lbm/hr ft]

$P_{r\ell}$ = liquid Prandtl number

To determine the steam side surface heat transfer coefficient in the superheater (area D) the following relation is applied (Reference 12).

$$h = 0.023 \frac{k_b}{D} \left[\frac{GD}{\mu_b}\right]^{0.8} \left[\frac{\mu C_p}{k}\right]_b^{0.4}$$

This is the same relation as was applied to the compressed liquid region (area A).

To determine the overall heat transfer coefficient for each area of the boiler the following relation is applied to each region in the boiler.

$$\frac{1}{U_s} = \frac{1}{\frac{A_g}{A_s} \varepsilon_g h_g} + \frac{1}{\frac{A_g}{A_s} h_{scg}} + \frac{D/2 \ \ln r_2/r_1}{k} + \frac{1}{h_s} + \frac{1}{h_{scs}}$$

A_g = boiler tube and fin outside surface area per inch of tube length [in^2/in.]

A_s = boiler tube inside surface area per inch of tube length [in.^{2}/in.]

ε_g = boiler tube surface effectiveness

h_g = boiler tube outside surface heat transfer coefficient [Btu/hr ft^2 °F]

h_{scg} = boiler tube outside surface fouling heat transfer coefficient [Btu/hr ft^2 °F]

D = boiler tube inside diameter [ft]

r_2/r_1 = ratio of boiler tube outside and inside radii

k = boiler tube metal thermal conductivity [Btu/hr ft °F]

h_s = boiler tube inside surface heat transfer coefficient [Btu/hr ft^2 °F]

h_{scg} = boiler tube inside surface fouling heat transfer coefficient

Once the overall heat transfer coefficient is determined for each section of the boiler (areas A, B, C, and D) the amount of inside surface area of the boiler tubes is determined using the following relation.

$$A = \frac{q}{U} \frac{\ln(\Delta T_a/\Delta T_b)}{\Delta T_a - \Delta T_b}$$

q = heat transfer rate [Btu/hr]

U = overall heat transfer coefficient [Btu/hr ft^2 °F]

$\dfrac{\Delta T_a - \Delta T_b}{\ln(\Delta T_a/\Delta T_b)}$ = log mean temperature difference at each area in the boiler [°F]

The total boiler tube inside surface area (A_T) is determined by adding the surface area computed for each section of the boiler. The length of boiler tube required in the boiler is computed using the following relation.

$$L = \frac{A_T}{\pi D_i}$$

L = total pipe length [ft]

D_i = boiler tube inside diameter [in.]

This value of tube length required to fulfill boiler heat transfer requirements is compared with tube length determined from diesel exhaust gas pressure differential. The boiler dimensions are adjusted to accommodate the amount of boiler tubing determined from the heat transfer analysis if the initially assumed tube arrangement did not have the same length of tubing as that determined from the heat transfer analysis.

CHAPTER 5

STEAM SYSTEM DESIGN

5.1 Machinery Components

The supply of steam from the diesel engine exhaust waste heat boiler will exceed shipboard steam demand, therefore, it becomes advantageous to convert the steams thermal energy into electrical energy for shipboard use. This can be accomplished using the arrangement of components shown in Figure 7. This figure illustrates the relationship of the various system components as well as steam extraction points and the points where the extracted steam is returned to the system. A diesel jacket water cooler is employed to increase the temperature of condensate supplied to the deairating feed tank (DFT) and extract some of the energy from the diesel jacket cooling water. Increasing the temperature of the condensate supplied to the DFT reduces the amount of steam required in the DFT to raise the outlet water temperature to 300°F. This temperature is required to reduce the possibility of sulfuric acid condensation and subsequent corrosion on boiler exhaust gas side heat transfer surfaces.

In order to perform an energy evaluation of the steam system (Figure 7) the following quantities are defined.

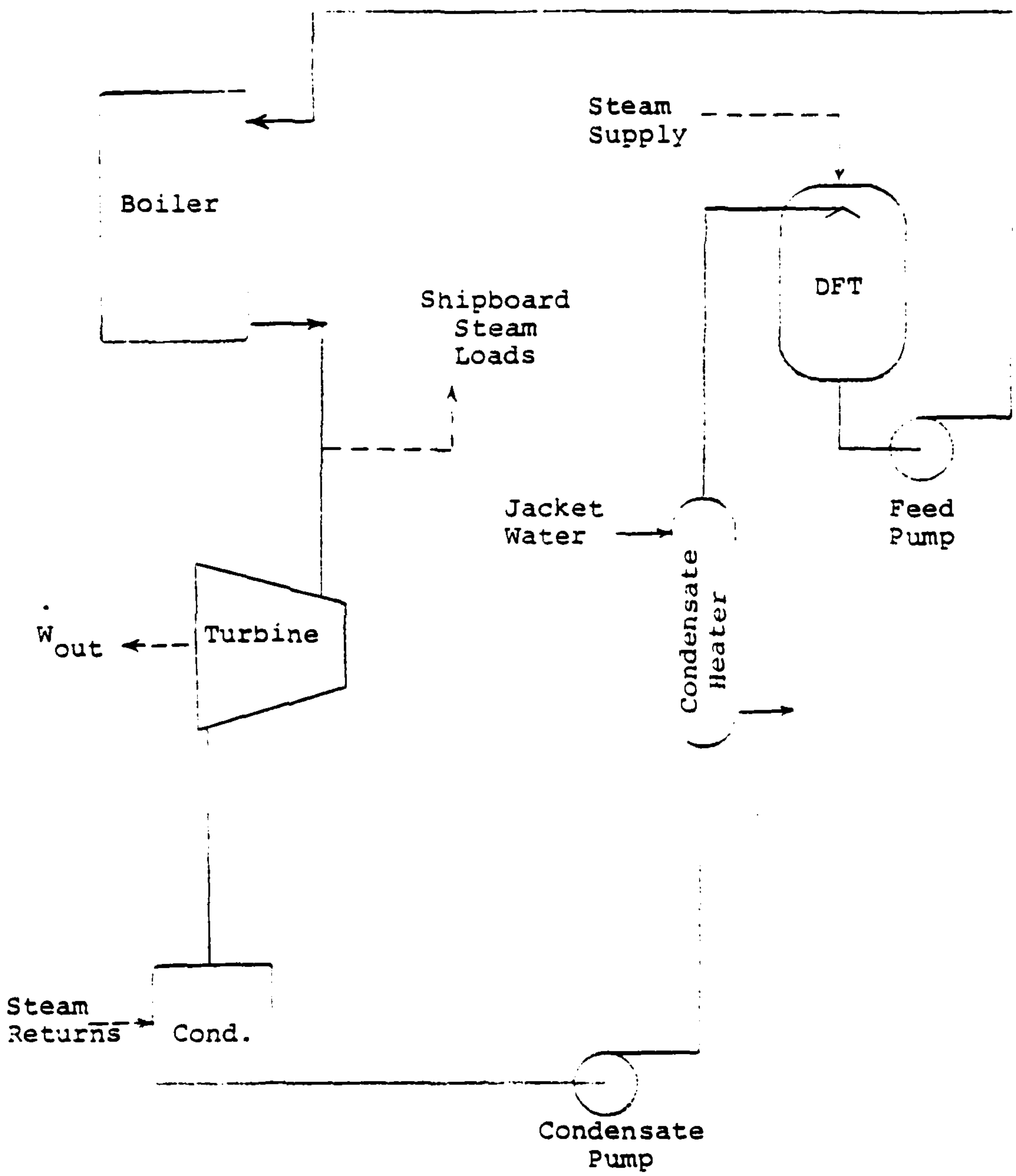

Steam
Supply
Boiler
DFT
Shipboard
Steam
Loads
Jacket
Water
Feed
Pump
$\dot{W}_{out}$
Turbine
Condensate
Heater
Steam
Returns
Cond.
Condensate
Pump

Figure 7

$\dot{m}_6$ = mass flow rate of steam from the boiler [lbm/hr]

$\dot{m}_a$ = mass flow rate of steam to shipboard steam loads [lbm/hr]

h_6 = enthalpy of steam leaving the boiler [Btu/lbm]

η_T = turbine efficiency

h_7 = enthalpy of turbine exhaust into the condenser [Btu/lbm]

T_7 = turbine exhaust temperature [°F]

P_7 = turbine exhaust pressure [psia]

$\dot{m}_R$ = mass flow rate of steam returns to the condenser [lbm/hr]

P_c = condensate pump discharge pressure [psig]

$\dot{m}_J$ = mass flow rate of diesel jacket cooling water [lbm/hr]

$T_{J_{in}}$ = inlet temperature of diesel jacket cooling water [°F]

$T_{J_{out}}$ = outlet temperature of diesel jacket cooling water [°F]

T_c = temperature of condensate exiting the condensate heater [°F]

h_c = enthalpy of condensate exiting the condensate heater [Btu/lbm]

$\dot{m}_s$ = mass flow rate of steam into the DFT [lbm/hr]

h_s = enthalpy of steam into the DFT [Btu/lbm]

$\dot{m}_{DFT}$ = mass flow rate of water exiting the DFT [lbm/hr]

h_{DFT} = enthalpy of water exiting the DFT [Btu/lbm]

P_F = discharge pressure of the boiler feed pump [psig]

To determine the mass flow rate of steam from the boilers the sum of the mass flows from each boiler is added. This flow rate equals $\dot{m}_6$.

To determine the power available from the turbine the following relation is used.

$$\dot{W}_{turbine} = \eta_T(\dot{m}_6 - \dot{m}_a)(h_6 - h_7)$$

The value of h_7 is determined once the values of T_7 and P_7 have been selected for the design. For design consideration the amount of steam entering the condenser is given by the following relation.

$$\dot{m}_{steam\ cond} = \dot{m}_6 - \dot{m}_a + \dot{m}_R$$

As previously stated it is necessary to preheat boiler feed water to at least 300°F to inhibit sulfuric acid corrosion of boiler gas side heat transfer surfaces. Heating is accomplishment in the condensate heater and the DFT. It becomes necessary to maintain condensate pressure above the saturation pressure for this temperature. Consequently, condensate pressure must be maintained at or above 55 psig. This pressure is selected as the condensate pump discharge pressure (P_c). To determine the discharge temperature of engine jacket cooling water from the condensate heater the following relation is used

$$T_{J_{out}} = T_{J_{in}} - \left[\frac{\dot{m}_6 - \dot{m}_a + \dot{m}_R}{\dot{m}_J}\right](T_6 - T_7)$$

The desired condensate discharge temperature is assumed and inserted into the previous relation. Because $\dot{m}_J$ is much larger than the mass flow rate of condensate a reasonable value is selected for $\dot{m}_J$ such that $\dot{m}_{condensate} \geq 1/10\ \dot{m}_J$.

In order to determine the steam supply into the DFT required to maintain condensate discharge temperature at 300°F the following relation is applied.

$$\dot{m}_s = \frac{\dot{m}_6\ h_{DFT} + (\dot{m}_6 - \dot{m}_s)h_c}{h_s}$$

For this relation the value of h_{DFT} is selected based upon a DFT water discharge temperature at or above 300°F. The value of steam enthalpy into the DFT is selected based upon an inlet pressure and temperature compatable with DFT operating parameters.

5.1.1 Machinery Component Sizing

The diesel main propulsor and ships service diesel generator waste heat boilers were sized using the design methodology presented in Chapter 4. This sizing was premised upon an operating profile of the main diesel engine operating at 50% of its maximum continuous rating and the ships service diesel generators operating at 75% of maximum continuous rating. For the purpose of sizing components in the steam

plant the maximum steam supply from the diesel engine waste heat boilers will be the design point. This design point will be defined by the amount of steam generated from the waste heat boilers with the main engine operating at 50% maximum continuous rating and two of the three ships service diesel generators each operating at 75% maximum continuous rating.

The turbine selected for use with diesel exhaust generated steam is the Terry Z-Line Solid Wheel Turbine, model 24Z. The turbine uses a single stage bucket wheel design for helical steam flow. The primary advantages of the turbine design are simplicity, compact size, and continued operation should steam flow carry entrained moisture into the turbine. In the operating range of steam flow the turbine has a 50% efficiency.

The condenser selected is a tube and shell crossflow heat exchanger with steam flow on the shell side and salt water coolant flowing through the tubes. In order to size the condenser for this application the following parameters are defined.

D inside tube diameter [in.]

N number of tubes

L tube length [feet]

g gravitational acceleration [ft/sec^2]

ΔP_c	pressure drop on tube side [psid]
f	friction factor
ρ_c	density of cooling water [lbm/ft^3]
V_c	velocity of cooling water [ft/sec]
A_c	cross sectional free flow area on tube side [ft^2]
$\dot{m}_c$	flow rate of cooling water [lbm/sec]
$\dot{q}$	heat transferred in the condenser [Btu/hr]
C_p	specific heat of cooling water [Btu/lbm °F]
C_c	$\dot{m}_c \times C_p$ [Btu/hr °F]
T_c	temperature difference between cooling water inlet and outlet [°F]
X	steam quality of inlet steam into the condenser
h_{fg}	enthalpy difference between saturated vapor of condensing steam and saturated liquid [Btu/lbm]

$\dot{m}_s$ mass flow of steam entering condenser [lbm/sec]

A_{HT} heat transfer area in the condenser relative to inside of tubes [ft^2]

Using the above parameters the following relationships are derived.

$$A_{HT} = \pi DNL$$

for: $$N = \frac{4A_c}{\pi D^2}$$

$$L = \frac{g \, \Delta P_c \, D}{2f \, \rho_c \, V_c^2}$$

$$A_{HT} = \frac{2 \, A_c \, g \, \Delta P_c}{f \, \rho_c \, V_c^2}$$

for: $$A_c = \frac{\dot{m}_c}{\rho_c \, V_c}$$

$$\dot{m}_c = q/C_c \, \Delta T_c$$

$$\dot{q} = \dot{m}_s \, h_{fg} \, X$$

$$A_{HT} = \left[\frac{2\ g\ \Delta P_c\ h_{fg}}{f\ \rho_c^{\ 2}\ V_c^{\ 3}\ C_c\ \Delta T_c}\right] \dot{m}_s\ X = K\ \dot{m}_s\ X$$

For similar condenser designs it is assumpd that condenser volume (V_c) is proportional to the condenser heat transfer area (A_{HT}).

$$\frac{V_{c_1}}{V_{c_2}} = \frac{K_1}{K_2}\ \frac{(\dot{m}_s\ X)_1}{(\dot{m}_s\ X)_2}$$

For similar condenser designs assuming the same tube side pressure differential (ΔP_c) the value of K_1/K_2 is approximately 1. The condenser size (V_{c_1}) can be reduced to the following expression.

$$V_{c_1} = \left[\frac{V_{c_2}}{(\dot{m}_s\ X)_2}\right] (\dot{m}_s\ X)_1$$

Using data from previous designs [15] the value in brackets may be evaluated for this design to provide the following relation.

$$V_c = 27.3\ \dot{m}_s\ X$$

This relation provides the condenser volume in ft^3 for $\dot{m}_s$ expressed in units of lbm/sec

In order to determine the size of the condensate heater using diesel engine waste heat supplied by the engine jacket cooling water the following analysis is presented. The relation of between jacket water and condensate flows is presented in Figure 8 .

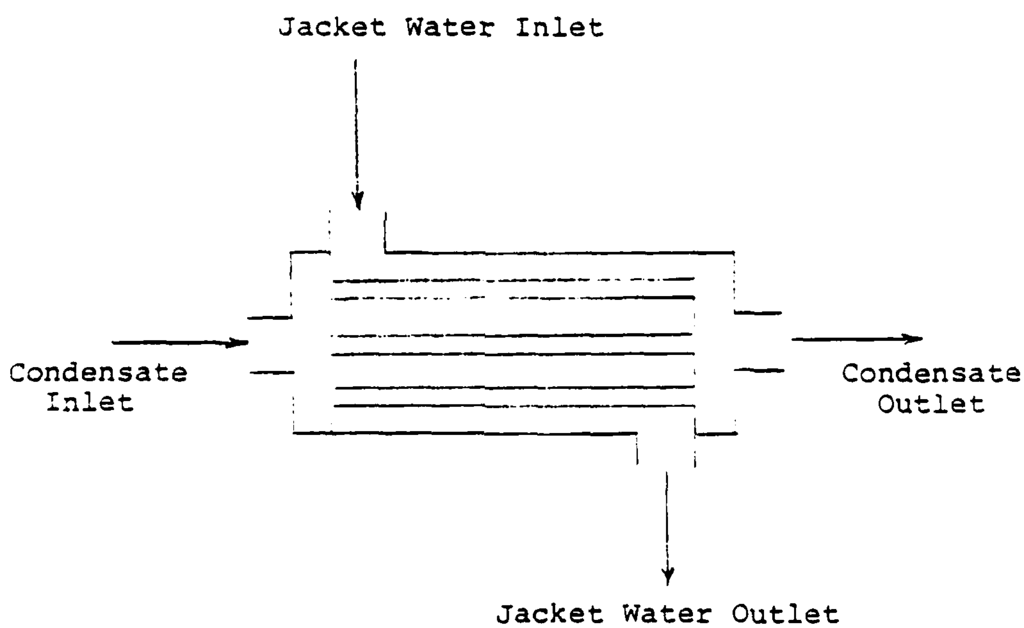


Figure 8

The heat exchanger employed is a shell and tube crossflow arrangement. Condensate is heated as it passes through the tubes and jacket water flows on the shell side of the heat exchanger. The method for determining required heat transfer area and ultimate sizing of the heat exchanger is presented (Reference 12).

ΔT_{LM} = log mean temperature difference [°F]

C_h = product of jacket water mass flow rate and specific heat [Btu/lbm °F]

C_c = product of condensate mass flow rate and specific heat [Btu/lbm °F]

$$\varepsilon = \frac{\Delta T_{condensate}}{T_{jacket\ in} - T_{condensate\ in}}$$

Using values determined for C_h, C_c, ΔT_{LM} and ε and curves provided in Reference 12 a value for F is determined

$$\Delta T_m = F \Delta T_{LM}$$

This is used to determine the tube inside surface heat transfer area.

$$A = \frac{q}{U \Delta T_m} \; [ft^2]$$

$$U = 400 \; \frac{Btu}{hr \; ft^2 \; °F} \; \text{(nominal value)}$$

The volume of the heat exchanger can be determined by specifying the inside tube diameter, the member of tubes desired and a tube arrangement. The total inside tube perimeter of the tube bundle (P) is computed and used to determine the length of the tube bundle.

$$L = \frac{A}{P} \; [ft]$$

To determine the size of the deaerating feed tank a parametric analysis is conducted using an existing design from another waste heat boiler configuration presented in Reference 16. The deaerating feed tank performs three functions in the steam generating system. The DFT removes air and noncondensable gases dissolved in solution, preheats the feed water, and acts as a surge volume for the system. Because the DFT acts as a surge volume the amount of water in the surge reservoir is assumed to be proportional to the size of the steam generating system. The size of the system is quantified by the rate of steam generation. Consequently, the ratio of surge volume (or the mass of water in the DFT)

to the rate of steam generation is assumed equal to a similar ratio for another waste heat recovery steam system. This ratio is presented below.

$$\left(\frac{\text{DFT water weight}}{\dot{m}_{steam}}\right)_{I} = \left(\frac{\text{DFT water weight}}{\dot{m}_{steam}}\right)_{II}$$

Using known values for the rate of steam generation and DFT surge capacity for another system [16] the relation becomes:

$$\text{DFT water weight} = 0.236\ \dot{m}_{steam}$$

DFT water weight [lbm]

$\dot{m}_{steam}$ [lbm/hr]

The surge volume is computed using the following relation.

$$\text{DFT water volume} = \text{DFT water weight} \times \text{water specific volume}$$

It is assumed that the surge volume is 2/3 of the total DFT volume. This permits calculations of total DFT volume required.

CHAPTER 6

CONCLUSIONS

The analysis presented has demonstrated a means of recovery of exhausted energy from main propulsion and ships service electrical power diesel engines. The most viable energy recovery is from diesel exhaust combustion gases. The amount of energy recovered from the main propulsion diesel operating at 50% maximum continuous rating and a ships service diesel engine operating at 75% maximum continuous rating is presented in Table 6.1.

	Main Propulsion Diesel Engine	Ships Service Diesel Engine
Net Energy Recovered [Btu/sec]	3328	810
Net Energy Used [Btu/sec]	1148	

Table 6.1

In a configuration of one ships service diesel engine and the main propulsion diesel engine sufficient energy is recovered to supply ships service steam loads and develop 337 KW

of electrical power assuming a generator efficiency of 0.95. This improves fuel efficiency by 4.3% when considering fuel consumption for the two engines.

To assess the improvement in fuel efficiency for various operating profiles the following engineering configurations are presented.

Configuration	Main Engine Operating	Number of Ships Service Engines Operating
A	no	1
B	yes	1
C	yes	2

Data relevant to heat recovery for these configurations is presented in Table 6.2.

	Configuration A	Configuration B	Configuration C
Net Energy Recovered [Btu/sec]	811	4140	4951
Net Energy Used [Btu/sec]	202	1148.7	1468
Electric Power Generated [KW]	0*	337	416
Percent Fuel Efficiency Improved [%]	3.5	4.3	4.5

Table 6.2

*All the steam produced by the waste heat boiler is used for shipboard steam loads consequently none is supplied for electric power generation.

The amount of fuel required by a naval surface combatant vessel is largely dictated by required endurance. By improving the fuel efficiency of the diesel engines using waste heat recovery it becomes possible to achieve the same endurance range for the vessel with less fuel. Consequently, less fuel storage volume is required. This allows volume reallocation or potential weight reduction of the vessel due to reduced fuel storage requirements.

The savings which can be derived from waste heat recovery amount to $128,000.00 per year. This assumes the vessel to be operating at sea four months per year in an engineering configuration of the main propulsion diesel at 50% maximum continuous rating and one ships service diesel operating at 75% maximum continuous rating.

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APPENDIX 1

WASTE HEAT RECOVERY BOILER DESIGN
(Sample Data)

Data presented uses the symbols employed in Chapter 4. The data presented is for the main propulsion engine, a Colt-Pielstick, 18 cylinder, type PC4 diesel engine, hereafter referred to as PC4. Data presented for a ships service electrical power diesel driven generator represents a Pielstick PA6V type 280, 12 cylinder diesel engine, hereafter referred to as PA6V.

Diesel Engine	PA6V	PC4
T_1 [°F]	115	115
T_2 [°F]	300	300
T_6 [°F]	600	600
T_{12} [°F]	707	692.6
$\dot{m}_G$ [lbm/sec]	11.4	46.74
$\dot{m}_W$ [lbm/hr]	2,315	11,556

P_2[psig]	300	300
P_6[psig]	300	300
η_p(is)	0.70	0.70
η_t(is)	0.50	0.50
C_{p_G}[Btu/lbm °F]	0.255	0.255
Q[Btu/sec]	810.92	3328.96
T_{pp}[°F]	39	24.4
t_m[in.]	0.045	0.045
ff_i[hr ft^2 °F/Btu]	0.001	0.001
ff_o[hr ft^2 °F/Btu]	0.01	0.01
A_A[ft^2]	184.75	710.41
A_B[ft^2]	236.19	1216.86

A_C [ft^2]	46.01	192.05
A_D [ft^2]	243.13	444.04
A_{TOTAL} [ft^2]	710.08	2563.36
TOTAL TUBE LENGTH [ft]	2059.12	6325.14
Number of Tube Rows	40	46
Tubes/Row	13	21
Tube Length [ft]	4.16	6.60
Boiler Height [ft]	11.0	12.75
Boiler Side Width [ft]	4.16	6.60

Data presented for the PA6V diesel represents calculations for a single engine operating at 75% maximum continuous rating. In the design of the waste heat boiler for this application the configuration which will govern boiler size will be when two PA6V diesel engines are operating at 75%

maximum continuous rating. Both engines will exhaust combustion gases into a single waste heat boiler. This will require a boiler capable of twice the heat recovery requirement of the single diesel engine boiler design. Consequently, twice the heat transfer surface area is required. This can be accomplished by doubling the area of the boiler normal to the combustion gas flow while maintaining the same height of the boiler. This will maintain the pressure differential across the boiler of the diesel exhaust gases within design specification (6 inches of water pressure differential).

The boiler specifications adjusted for tandem PA6V engine operation are as listed below.

Number of tube rows	46
Tubes/row	29
Tube Length [ft]	9.50
Boiler Height [ft]	12.75
Boiler Side Width [ft]	9.33

APPENDIX 2

STEAM SYSTEM COMPONENT DESIGN
(Sampler Data)

A. Condenser

The condenser is sized using the methodology presented in Chapter 5. The condenser size is determined by the maximum amount of steam generated by the system. This occurs when the main propulsion diesel operates at 50% maximum continuous rating and 2 of the ships service diesel engines are each operating at 75% maximum continuous rating. The amount of steam produced in this plant operating configuration is reduced by the steam provided to the DFT, as this steam bypasses the condenser. Using the relation developed in Chapter 5 the condenser volume is determined.

$\dot{m}_G = 4.2$, the mass flow rate of steam into the condenser [lbm/sec]

$X = 0.90$, the quality of the steam into the condenser

$V_c = (27.3)\dot{m}_x X$, condenser volume [ft^3]

$V_c = 103.2 \text{ ft}^3$

B. Condensate Heater

The condensate heater is designed using the methodology presented in Chapter 5. Symbols used to design the heat exchanger are as presented in Chapter 5.

$\Delta T_{LM} = 34.1\ °F$

$C_h/C_c = 10$

$\varepsilon = 0.79$

$F = 0.95$

$\Delta T_m = 32.4\ °F$

$A = 61\ ft^2$

Using a tube outside diameter of 0.625 inches and a tube bundle of 64 tubes gives a tube length of 5.8 ft. The tube bundle dimensions becomes 5.8 feet long with a circular bundle 1 foot in diameter.

C. Deaerating Feed Tank

The DFT is designed using the methodology presented in Chapter 5. The DFT size is determined using the following relations.

$$\text{DFT water weight} = (0.236)\ \dot{m}_{steam}$$

$$= 4055 \text{ lbm}$$

$$\text{DFT water volume} = 71 \text{ ft}^3$$

$$\text{DFT total volume} = 107 \text{ ft}^3$$

APPENDIX 3

STEAM SYSTEM COMPONENT ARRANGEMENTS

Steam system components are arranged in a representative naval combatant hull form in Figures 9 and 10. Figure 9 shows the arrangement of the exhaust gas boiler relative to the three ships service diesel generators. Also located in this compartment are the steam system components. This space was selected for locating the steam system to allow minimal impact during pierside auxiliary steaming. This would allow waterstanders to monitor the steam system while ship service generators are operating.

The arrangement of the main propulsion diesel engine and associated exhaust gas waste heat boiler is presented in Figure 10. In both figures the size of components relative to the diesel engines gives an indication of the ability to integrate the diesel exhaust waste heat recovery system with a diesel engine arrangement.

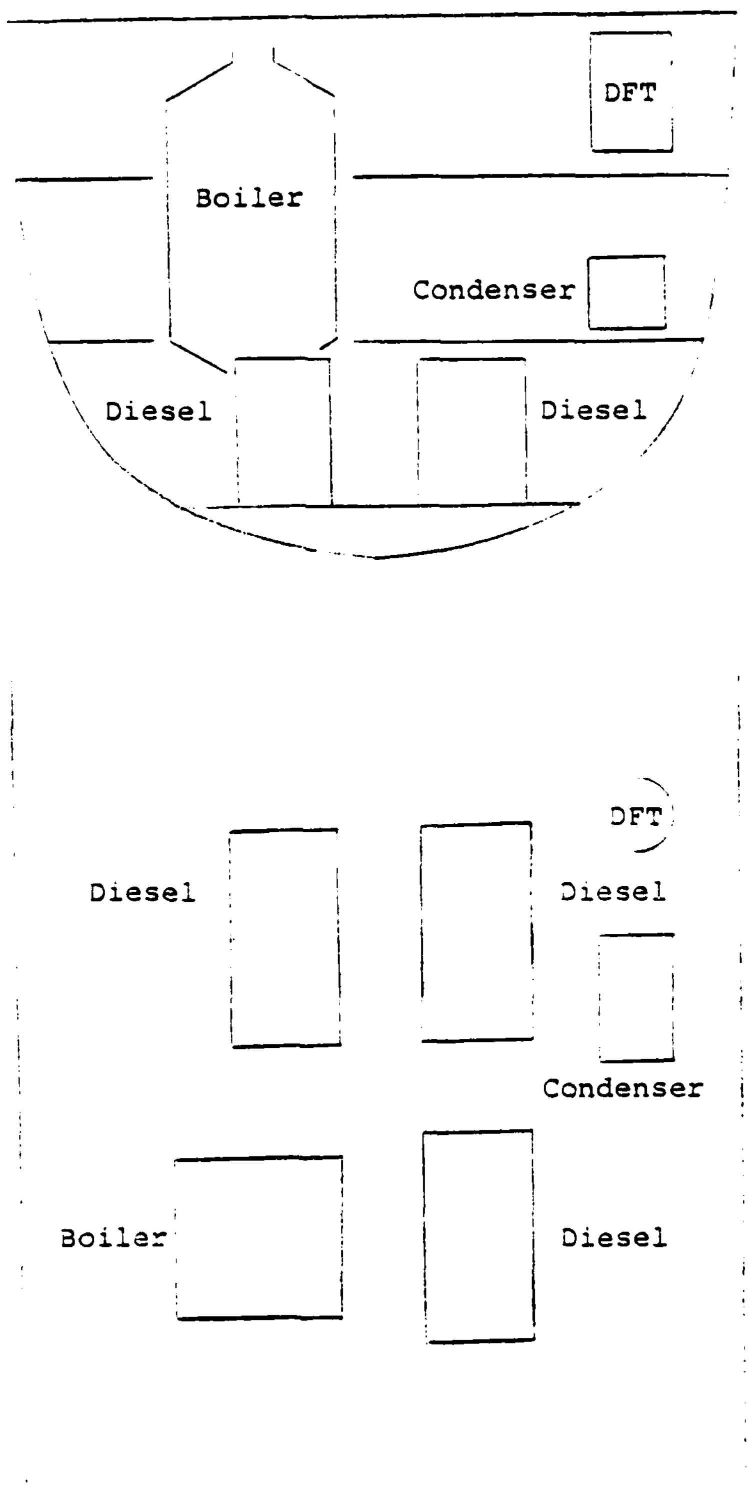
DFT
Boiler
Condenser
Diesel
Diesel
DFT
Diesel
Diesel
Condenser
Boiler
Diesel

Figure 9

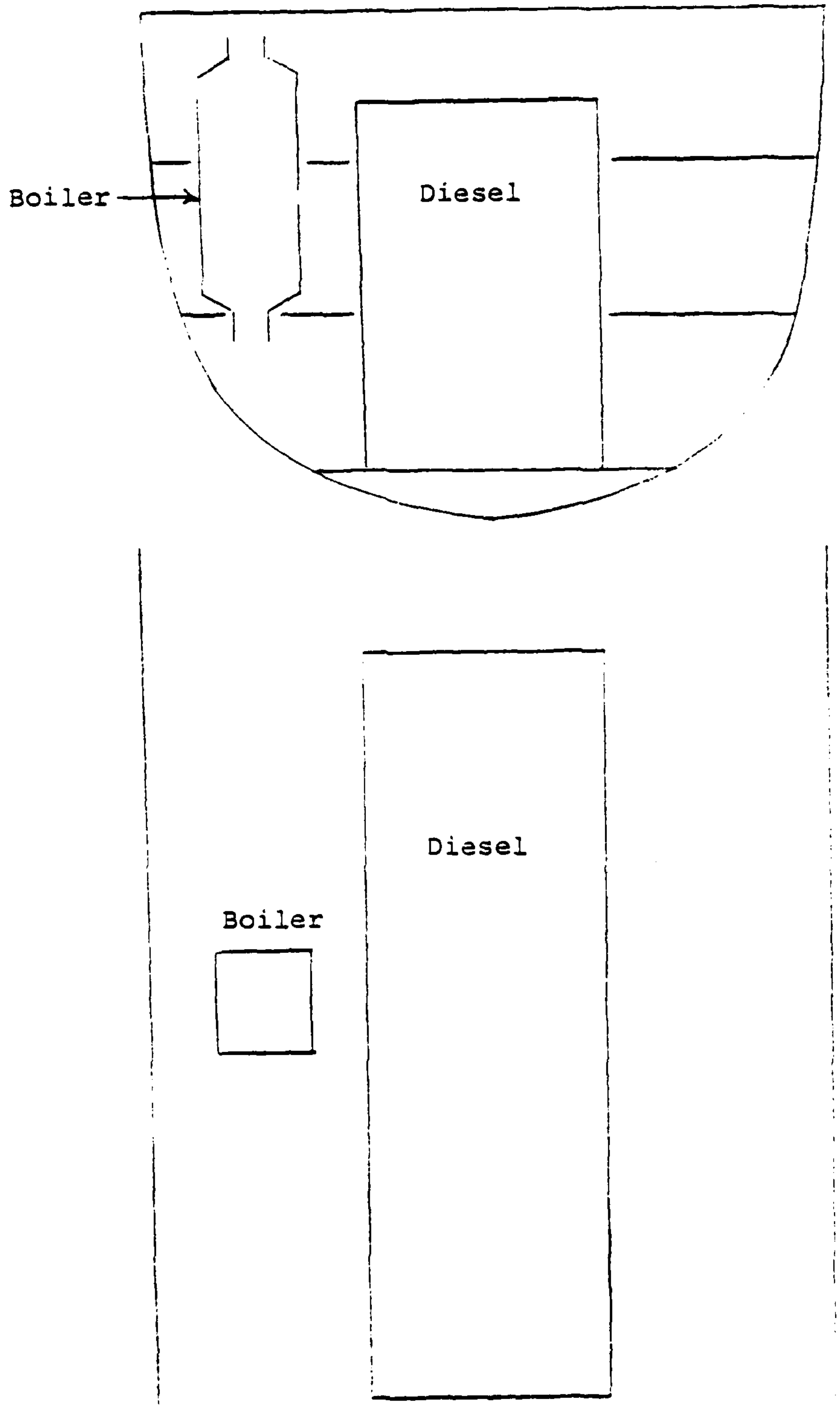

Figure 10

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